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Influence of the working fluid thermophysical parameters variation on the gas turbine cycle performance

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Abstract. The thermodynamic analysis of effect of the syngas combustion products (working fluid) thermophysical parameters variation on the economic and energy parameters of the gas turbine cycle is described. The thermodynamic analysis of Brayton cycle performance on various compositions of working fluid within a selected range of control parameters variation is presented. A gas turbine working fluid database was generated based on the analysis of flow diagram and operating regime of the production and design alternative gas-fired CCPP. The relationship between the cycle thermodynamic (pressure and temperature) and thermophysical parameters of the working fluid, which ensures gas turbine cycle maximum work, is derived analytically and supported by the well-known actual data.

1. Introduction

Increasing the economic and energy parameters of a gas turbine unit, both in a simple cycle and in a combined cycle, is a crucial task at the present day. The main parameters characterizing the economical and energy efficiencies of the gas turbine cycle are gas turbine cycle actual efficiency η_i and actual useful work l_0^{act}.

The main method to increase economic and energy parameters of the gas turbine unit is to increase pressure P_3 and temperature t_3 in the gas turbine inlet flow (gas turbine working fluid). The leading manufacturers have developed their own ranges of parameters (P_3 and t_3) for traditional (natural gas) fuel-fired gas turbines, which correspond to optimal conditions of the plant operation. For the achievement of the optimal economic and energy design parameters, the working fluid is subjected to parameter correction when standard fuel-fired gas turbine is converted to non-project alternative fuel-fired.

The flow diagram and operating regime of the production and design alternative gas-fired CCPP (e.g. syngas, artificial gas, blast-furnace gas, coke-oven gas etc.) analysis revealed an existence of some methods of gas turbine working fluid composition correction. The known methods of the gas turbine working fluid composition correction are listed below:

- gas fuel or oxidizer (air or O_2) dilution by nitrogen, water or CO_2;
- enrichment of gas fuel (by coke-oven or natural gas) and oxidizer (by O_2);
- removal of the least energy-intensive component CO_2 by pre-combustion CCS;
- changing the thermal regime for gas fuel and oxidizer preparation.

Optimization of the thermodynamic and design data of GTU is necessary when developing new gas-turbine cycles (e.g., Oxy-fuel [2], Zecomix cycle [3], Allam cycle [4]).

The optimal operation conditions of GTU on the basis of new working fluids (H_2O, CO_2) change and cycle thermodynamic parameters should be corrected.

The paper is dedicated to the thermodynamic investigation of the influence of gas turbine working fluid (alternative gas fuels combustion products) thermophysical parameters variation on the economic and energy parameters of the simple gas turbine cycle.

2. Method

The investigation of the influence of gas turbine working fluid thermophysical parameters variation on the gas turbine cycle performance was carried out in the range of control parameters presented in table.1.

Table 1. Range of control parameters variation

Parameter	Dimension	Values	
		min	max
P_3	MPa	1.4	30.0
t_3	°C	1100	1500
H_2O	vol.%	0	100
CO_2	vol.%	0	100
k	-	1.170	1.669
μ	kg/kmol	4	131
c_{pm}	kJ/(kg·K)	0.158	5.204

The choice of gas turbine inlet pressure P_3 and temperature t_3 (initial thermodynamic parameters of working fluid) is determined by currently accepted for existing and development gas turbine classes, modified for operation on alternative gas fuels [1, 2, 3, 4].
The range of changes in the heat capacity ratio k, molar mass μ and specific heat c_{pm} and $c_{p\mu}$ corresponds to the working fluids parameters used in the practice of unclosed and closed gas turbine cycles and consisting of pure fluid and their combinations (table.2).

Table 2. The characteristics of the gas turbine working fluids
(in the accepted temperature range t_3=1100÷1500 °C and pressures P_3=1.4÷30 MPa)

Fluid	μ, kg/kmol	k	c_{pm}, kJ/(kg·K)	$c_{p\mu}$, kJ/(kmol·K)	Fluid type	Application
Helium (He)	4	1.666	5.204	20.816	G	Closed cycle
Steam (H_2O)	18	1.255÷1.264	2.210÷2.275	39.785÷40.941	H	Closed cycle, Unclosed cycle
Natural gas combustion products (NGC)	28.38÷ 28.48	1.324÷1.335	1.152÷1.184	32.817÷33.617	A	Unclosed cycle
Nitrogen (N_2)	28	1.343÷1.352	1.125÷1.142	31.486÷31.981	G	Closed cycle
Air	28.84	1.339÷1.347	1.107÷1.124	31.934÷32.421	A	Closed cycle
Carbon dioxide (CO_2)	44	1.191÷1.200	1.157÷1.179	50.890÷51.859	C	Closed cycle, Unclosed cycle
Xenon (Xe)	131	1.666	0.158	20.750	G	Closed cycle

A variety of working fluids compositions formed on basis of given in table 2 fluids, enlarged can be divided into four types, allocated according to the technology of their production and/or application:

- **Type A** (from “Air”) – the working fluids as being the air combustion products of natural and alternative gases. It is used in unclosed gas turbines cycle both production and developed natural gas-fired CCPP, IGCC and artificial gas-fired CCPP [5, 6];
- **Type H** (from “Hydrogen”) – the working fluids as being the O_2-H_2O combustion products of syngas derived from coal by O_2-H_2O conversion and pre-combustion CCS. It is used in the developments of semi-closed gas turbine cycle and steam-turbine cycle [2, 4];
- **Type C** (from "Carbon") – working fluids as being the O_2-CO_2 combustion products of syngas derived from coal by O_2-CO_2 conversion. It is used in advanced developments of a semi-closed cycle gas turbine cycle as part of Oxy-fuel power plants [1, 3];

- **Type G** (from "Gas") – working fluids as being special fluids (He/Xe/N_2/Ar mixtures) used in low-power closed gas turbine cycle also known as High Temperature Reactor Helium Gas Turbine (HTR-GT) [7].

Working fluids types A, H and C with a molar mass μ in the range 18÷44 kg/kmol, specific molar heat capacity $c_{p\mu}$ in the range 31÷52 kJ/(kmol·K) and adiabatic exponent k in the range 1.191÷1.347 will be relevant for heavy-duty gas turbine unit as a part of alternative gas-fired CCPP.
Working fluids type G of low-power closed gas turbine cycle is characterized by the following parameters: μ=4÷131 kg/kmol, $c_{p\mu}$=20÷21 kJ/(kmol·K), k=1.666.

3. Result

In according with the first law of thermodynamics, the ideal work of the gas turbine l_{T} is equal to the difference between inlet and outlet enthalpy of working fluid (1):

$$l_{\mathrm{T}} = i_3 - i_4 = \mathrm{c}_{pm,3} \cdot T_3 - \mathrm{c}_{pm,4} \cdot T_4 \tag{1}$$

i_3, i_4 – working fluid specific enthalpy at the gas turbine inlet and outlet, kJ/kg; T_3, T_4 – working fluid temperature at the gas turbine inlet and outlet, K.
In the ideal-gas approximation with constant mass specific heat $\mathrm{c}_{pm,3} = \mathrm{c}_{pm,4}$ (prompting suggestions that the heat capacity ratio $k = \mathrm{c}_{pm}/\mathrm{c}_{pv}$ is independent of the parameters *P* and *T*), expression (1) for the working fluid will be rewritten:

$$\begin{gathered} l_{\mathrm{T}} = c_{pm,3} \cdot T_1 \cdot \xi \cdot \varphi_{\mathrm{T}} \\ \varphi_{\mathrm{T}} = 1 - \pi^{-m} = l_{\mathrm{T}}/\left(\mathrm{c}_{pm} \cdot T_3\right) = \eta_t < 1 \\ \xi = T_3/T_1; \quad m = (k-1)/k = R_\mu/c_{p\mu} < 1 \end{gathered} \tag{2}$$

φ_{T} – dimensionless gas turbine expansion work, numerically equal to the fraction of the working fluid enthalpy which convert to the work l_{T} (cycle thermal efficiency η_t); ξ – working fluid heat rate in the gas turbine cycle; R_μ =8.314 kJ/(kmol·K) – universal gas constant, equal to the 1 kmol of ideal gas specific expansion work in the isobaric process at temperature change in 1K; π – compression ratio in the compressor; m – fraction of working fluid thermal energy which convert to the work l_{T}.
Calculation of the compressor ideal work l_{C} under similar restrictions is carried out according to the formula (3):

$$\begin{gathered} l_{\mathrm{C}} = i_2 - i_1 = \mathrm{c}_{pm,2} \cdot T_2 - \mathrm{c}_{pm,1} \cdot T_1 = \mathrm{c}_{pm,1} \cdot T_1 \cdot (\pi^m - 1) = \left(\mathrm{c}_{pm,1} \cdot T_1\right) \cdot \varphi_{\mathrm{C}} \\ \varphi_{\mathrm{C}} = l_{\mathrm{C}}/\left(\mathrm{c}_{pm,1} \cdot T_1\right) = \pi^m - 1; \qquad \mathrm{c}_{pm,1} = \mathrm{c}_{pm,2} \end{gathered} \tag{3}$$

i_1, i_2 – compressible fluid specific enthalpy at the compressor inlet and outlet, kJ/kg; T_1, T_2 – compressible fluid temperature at the compressor inlet and outlet, K; φ_{C} – dimensionless compressor work.
Gas turbine and compressor ideal works ratio:

$$\psi = \frac{l_{\mathrm{T}}}{l_{\mathrm{C}}} = \frac{c_{pm,3}}{c_{pm,1}} \cdot \xi \cdot \frac{\varphi_{\mathrm{T}}}{\varphi_{\mathrm{C}}} \tag{4}$$

The heat and mass balances analysis of gas turbine revealed that when gas turbine operated at working fluid type G in a closed cycle or types H and C in a semi-closed cycle, the expression $c_{pm,3}/c_{pm,1} = 1$ is maintained with good accuracy for working fluid listed in table 2.
When gas turbine operated at working fluid type A, the divergence of mass specific heat capacities for advanced gas turbine in range of:

$$c_{pm,3}/c_{pm,1} = 1 + (0{,}1 \div 0{,}19) \tag{5}$$

Thus, in a first approximation, accept that the ratio of mass specific heat $c_{pm,3}/c_{pm,1}$ equal to 1, i.e. $c_{pm,3} = c_{pm,1} = c_{pm}$. Then the ideal works ratio of the gas turbine l_{T} and compressor l_{C} will be uniquely depend on working fluid superheat degree ξ_{T} in the gas turbine combustion chamber:

$$\psi = l_{\mathrm{T}}/l_{\mathrm{C}} = \xi \cdot \varphi_{\mathrm{T}}/\varphi_{\mathrm{C}} = \xi \cdot \pi^{-m} = T_3/T_2 = \xi_{\mathrm{T}} \quad (6)$$

The actual work of the gas turbine cycle is a function of two thermophysical parameters – a relatively slightly varying c_{pm} and a due to μ significantly varying $c_{p\mu}$.

For simplicity the thermodynamic analysis, accept that compressor and gas turbine flow rates are the same, and the gas turbine is a closed cycle. Then gas turbine cycle actual work l_0^{act} will be determined from the expression:

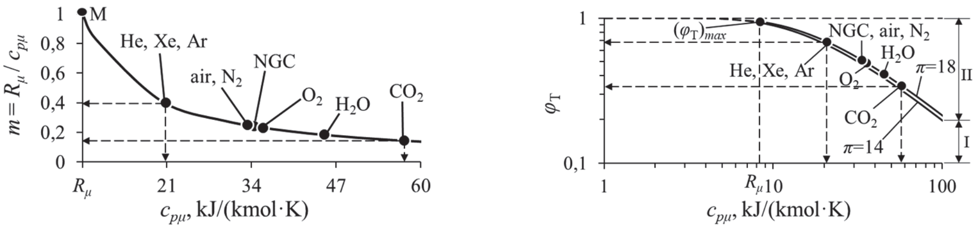
$$l_0^{act} = l_{\mathrm{T}} \cdot \eta_{is,\mathrm{T}} - l_{\mathrm{C}}/\eta_{is,\mathrm{C}} = c_{pm} \cdot T_1 \cdot (\pi^m - 1) \cdot \left(\eta_{is,\mathrm{T}} \cdot \xi_{\mathrm{T}} - \eta_{is,\mathrm{C}}{}^{-1}\right) = c_{pm} \cdot T_1 \cdot \tilde{l}_0^{act} \quad (7)$$

where $\tilde{l}_0^{act} = l_0^{act}/\left(c_{pm} \cdot \mathrm{T}_1\right)$ – dimensionless actual work of the gas turbine cycle;
$\eta_{is,\mathrm{T}}$ – gas turbine isentropic efficiency; $\eta_{is,\mathrm{C}}$ – compressor isentropic efficiency.

Expression (7) considers compressor and gas turbine frictional losses, cooling air losses and other irreversible losses. Compared to l_0^{act}, the actual efficiency η_i of a simple gas turbine cycle depends only on $c_{p\mu}$:

$$\eta_i = \frac{l_0^{act}}{q_1} = \frac{c_{pm} \cdot \mathrm{T}_1 \cdot (\pi^m - 1) \cdot \left(\eta_{is,\mathrm{T}} \cdot \xi_{\mathrm{T}} - \eta_{is,\mathrm{C}}{}^{-1}\right)}{c_{pm} \cdot \mathrm{T}_1 \cdot \pi^m \cdot (\xi_{\mathrm{T}} - 1)} = \frac{(\pi^m - 1) \cdot \left(\eta_{is,\mathrm{T}} \cdot \eta_{is,\mathrm{C}} \cdot \xi_{\mathrm{T}} - 1\right)}{\pi^m \cdot \eta_{is,\mathrm{C}} \cdot (\xi_{\mathrm{T}} - 1)} \quad (8)$$

one can see from the equation (2) that the lower the working fluid molar specific heat $c_{p\mu}$ the more thermal energy of the working fluid converts to the work l_{T}. Starting with the limit value m =1, with rising $c_{p\mu}$ the parameter m will decrease in terms of hyperbolic law (figure 1):



a) b)

Figure 1. Parameters m (a) and φ_{T} (b) as a functions of molar specific heat $c_{p\mu}$: I – fraction of the working fluid enthalpy, convert to the work l_{T}; II – fraction of the working fluid enthalpy, transmitted to the heat-recovery steam generator

Realization of the dependency of figure 1a is limited. The conditional actual values of parameter m for NGC (natural gas combustion products) and for single fluid, which are possible components of the gas turbine working fluid, are plotted by means of points in figure 1a. The actual values of parameter m for the investigational working fluid are in the range of 0.4>m>0.15.

The parameter φ_{T} as a function of molar specific heat $c_{p\mu}$ at a temperature of t_3 =1400 °C and a pressure of P_3 =1.8 MPa is represented in figure 1b. For selected working fluids, the parameter φ_{T} is ~35÷70% of the limit value, reaching $(\varphi_{\mathrm{T}})_{max}$ =0.93÷0.95 for advanced gas turbines with P_3 =1.4÷1.8 MPa.

The value $(1 - \varphi_{\mathrm{T}})$ is a fraction of the working fluid enthalpy, transmitted to the CCPP bottoming cycle (region II). The minimum transfer of the working fluid energy into the bottoming cycle $(1 - \varphi_{\mathrm{T}})$ =0.315 is for monatomic (He, He, Ar) working fluids; the maximum – for triatomic working fluids: $(1 - \varphi_{\mathrm{T}})$ =0.588 for H_2O and $(1 - \varphi_{\mathrm{T}})$ =0.658 for CO_2.

The gas turbine work l_{T} as a function of thermophysical (μ, c_{pm}) parameters of the working fluid is represented in figure 2. Apparently, with an increase molar mass μ, the gas turbine work l_{T} decreases sharply, while an increase c_{pm} leads to an increase l_{T}. The points M characterizes the limiting value of the working fluid molar mass for a specific value of the c_{pm} where $c_{p\mu} = R_{\mu}$.

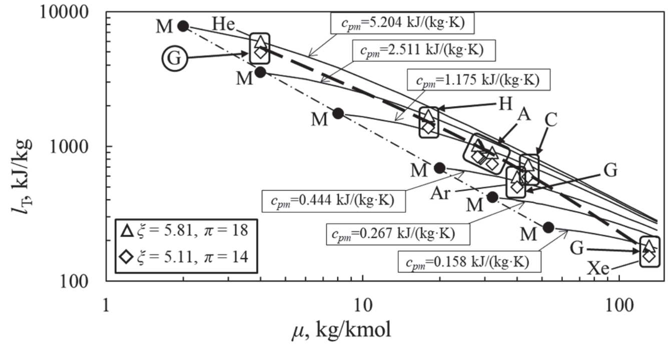


Figure 2. Work l_{T} as a function of thermophysical parameters (μ, c_{pm}) of the working fluid

As the $c_{p\mu}$ changes, the ideal dimensionless work $\tilde{l}_0^{act}$ reach a maximum and tending to zero at small and large values of $c_{p\mu}$ (figure 3a). Actual economic (η_i) and energy (q_1, l_0^{act}) parameters of as gas turbine cycle as a function of working fluid molar mass μ at c_{pm} =1 kJ/(kg·K), π =18, ξ =5.81, $\eta_{is,\mathrm{C}}$ =0.86 and $\eta_{is,\mathrm{T}}$ =0.88 are presented in figure 3b:

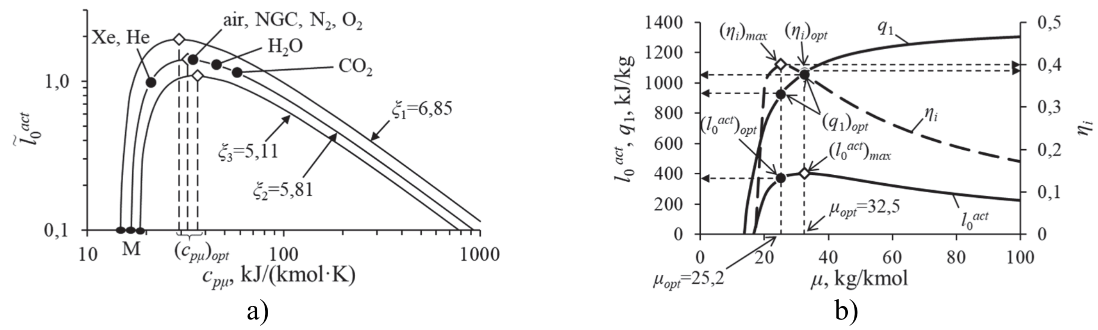


Figure 3. Parameter $\tilde{l}_0^{act}$ as a function of $c_{p\mu}$ (a) and parameters η_i, q_1, l_0^{act} as a function of μ (b) at c_{pm} =1 kJ/(kg·K)

The analysis of the data provided in figure 3, shows that $\left(c_{p\mu}\right)_{opt}$ with good accuracy coincides with $c_{p\mu}$ of NGC at pre-set values π =18, ξ =5.81. Under given parameters, the maximum actual work of the gas-turbine cycle corresponds μ_{opt} =32.5 kg/kmol, and the maximum actual efficiency of the cycle – μ_{opt} =25.2 kg/kmol. Such coincidence is not surprising as these parameters specially were selected by the producer for natural gas combustion in air.

At c_{pm} =2 kJ/(kg·K), the corresponding values of μ_{opt} will be 16.3 and 12.6 kg/kmol.

Generally, the optimal value of $\left(c_{p\mu}\right)_{opt}$, corresponding to the maximum value $\tilde{l}_0^{act}$, is found by the standard procedure for determining a function of one variable extremum:

$$\left.\frac{d\tilde{l}_0^{act}}{dc_{p\mu}}\right|_{c_{p\mu}=\left(c_{p\mu}\right)_{opt}} = \frac{d}{dc_{p\mu}}\left(\xi \cdot \left(1 - \frac{1}{\pi^{R_\mu/c_{p\mu}}}\right) \cdot \eta_{is,\mathrm{T}} - \left(\pi^{R_\mu/c_{p\mu}} - 1\right)/\eta_{is,\mathrm{C}}\right) = 0 \tag{9}$$

The solution to equation (9) is:

$$\left(c_{p\mu}\right)_{opt} = 2 \cdot R_\mu \cdot \frac{\ln \pi}{\ln[\eta_{is,\mathrm{C}} \cdot (\xi - \eta_{is,\mathrm{T}})]} \tag{10}$$

follows from the equation (10) that each pair of π and ξ corresponds to its optimal molar specific heat $(c_{p\mu})_{opt}$ of the working fluid; and inversely, each composition of the working fluid has its own optimal pair of thermodynamic parameters (figure 4a).

Conspicuous is the fact that the wide range of changes in the optimal π for an order of magnitude narrower range of ξ, necessary for the organization of gas turbine work on the investigational fluids. The dominant role of pressure ratio π specifies the direction of the theoretical cycles developments and constructive decisions towards increase in operating pressure.

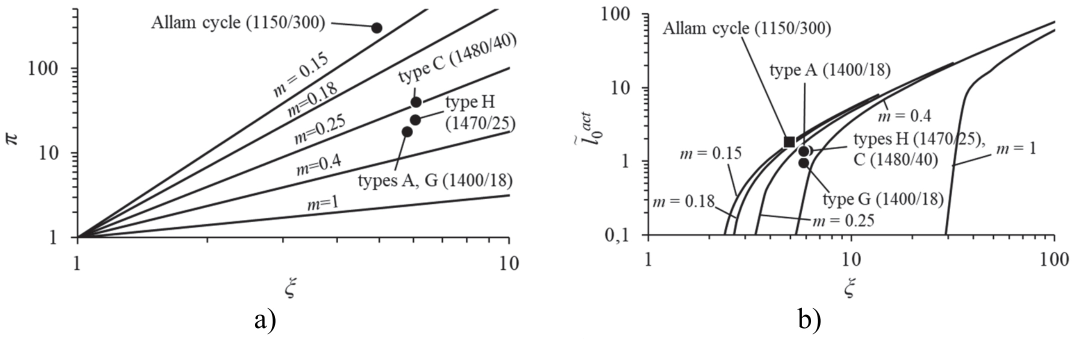


Figure 4. Optimal and actual π (a) and parameter $\tilde{l}_0^{act}$ (b) as a function of ξ for working fluids with different m (points are applied to the actual parameters of the turbine, when operating in different fluids).

The optimal cycle work found for the conditions shown in figure 4a is presented in figure 4b. The points plotted on the chart reveal the trend of the optimal actual cycle work, depending on the type of working fluid.

Conclusions

Variation of working fluid composition during the gas turbines operation on gas mixtures with different composition will significantly affect the gas turbine cycle actual work l_0^{act} and efficiency η_i.

The investigated parameters l_0^{act} and η_i regardless of gas turbine cycle type (closed, unclosed) and oxidizer type (air, oxygen) with a molar specific heat $c_{p\mu}$ variation reached a local maximum.

The position $(l_0^{act})_{max}$ is determined by working fluid molar specific heat $(c_{p\mu})_{opt}$.

For each type of working fluid, there is its own optimal π and ξ ratio, which can deviate the standard values considerably. With increase molar specific heat $c_{p\mu}$, the actual gas turbine cycle optimal process pressure increases in power law:

$$\pi_{opt} = [\eta_{is,\mathrm{C}} \cdot (\xi - \eta_{is,\mathrm{T}})]^{c_{p\mu}/(2 \cdot R_\mu)}$$

where $c_{p\mu}/(2 \cdot R_\mu) = 0{,}5 \cdot m$

When standard fuel-fired gas turbine converts to non-project alternative fuel-fired, the working fluid should be subjected to a corresponding parameter correction to achieve the optimal gas turbine energy parameters as included in the design.

When developing a new technological concept using non-standard working fluids, the thermodynamic parameters of the newly developed gas turbine should correspond to the new working fluid.

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